Interim Report on Preliminary Engineering Design Study for SNAP Secondary Mirror Support Structure

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Executive Summary

Finite element modal analyses have been performed to evaluate the suitability of a tripod design for the SNAP Secondary Mirror Support Structure. Specifics of the design were varied and analyzed to determine the design's sensitivity to these variations. Stress and buckling analyses under quasi-static acceleration loading were done on a promising design configuration to demonstrate the large strength margins that result from designing for high stiffness.

On-orbit performance goals include the ability to maintain a precise and well-controlled spatial relation between the Secondary Mirror Assembly and the Optical Bench, subject to thermal variations and the influence of attitude control and other spacecraft accelerations. This goal is to be met with a minimum of obscuration of the Primary Mirror and with a minimum of weight. Additionally, the support structure must survive the launch environment.

Specifically, the dynamic influences on mirror positioning are minimized and launch loads are tolerated by keeping the support structure natural frequency above 60-90 hertz. Thermal distortion effects are minimized by employing materials with near-zero coefficients of thermal expansion, possibly in combination with active thermal control of the structure. Primary mirror obscuration is minimized by making the projected width of the tripod legs as small as practical. Weight may be traded for stiffness to some extent, with the optimal balance to be determined subject to refined spacecraft requirements.

A possible design configuration meeting the goals of this study is illustrated in Figure 1. The material used throughout is a quasi-isotropic layup of carbon fiber/cyanate ester, with boron fibers added to attain near zero coefficient of thermal expansion. This design employs 1-1/2" x 10" hollow, rectangular legs, with 0.125" thick walls. Each leg is buttressed at its base with three flat plates 0.1" thick. This design weighs 157 pounds including the Secondary Mirror Assembly. It has a fundamental natural frequency of 61 hertz (in a twisting mode), and its lowest translational natural frequency is 95 hertz. Its launch load safety factors are greater than 10 for strength and greater than 30 for buckling. It obscures less than 3 percent of the primary mirror.

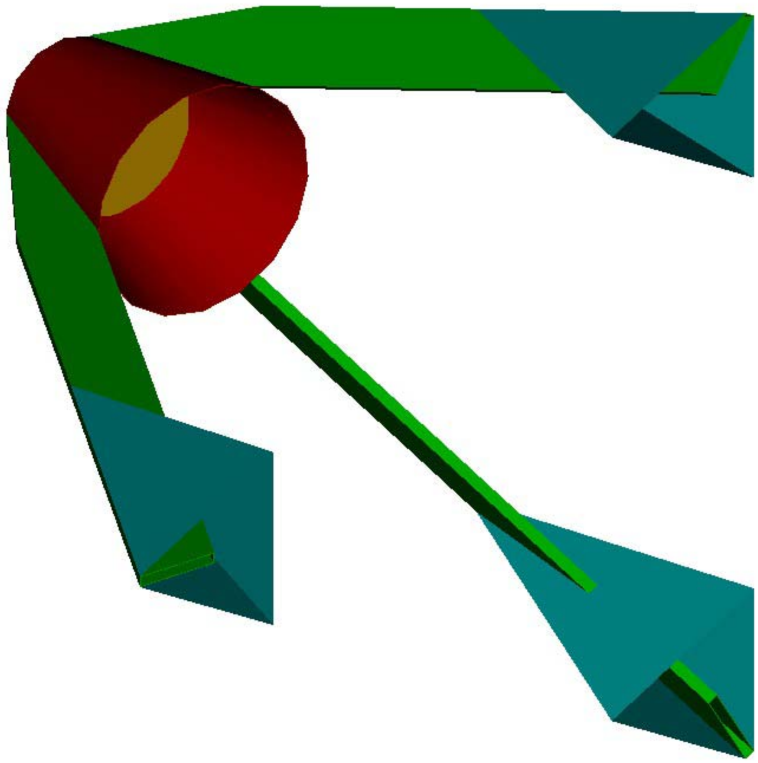

Figure 1-Design configuration

Description of the model

Several general layouts for the Secondary Mirror Support Structure have been previously considered. These include 3-, 4-, 6-, and 8-legged designs, either with the legs running all the way from the Optical Bench to the Secondary Mirror Assembly or with an intermediate cylindrical ‘elephant stand’ assembly constructed outside the Primary Mirror. For simplicity of fabrication and assembly, it is preferable to have a minimum number of legs and omit the ‘elephant stand’, if possible. Therefore, this study focuses exclusively on a tripod design mounted directly to the Optical Bench to evaluate if this simplest configuration promises satisfactory performance.

Figure 1 illustrates the model. Three legs (either hollow, rectangular box sections or flat plate sections) run from the Optical Bench to a representation of the Secondary Mirror Assembly. The Optical Bench end of each leg is buttressed by three flat plates that extend as far along the leg as possible without entering the field of view of the Primary Mirror. All the model’s nodes at the base of each leg and at the bottom of all the buttressing plates are fixed in six axes, representing the Optical Bench as perfectly rigid.

This study is not intended to detail the design of the Secondary Mirror Assembly itself, so it is modeled as a very simple construction and is made very stiff to minimize its effect on the support structure, other than by its mass. The Secondary Mirror Assembly is modeled as the surface of a truncated cone, occupied by two flat discs, one at the end farthest from the Primary Mirror, and one 11.1" from the first. The truncated cone is 30.6" long with diameters at the ends of 17.1" and 31.1". The larger end is 75.8" from the mounting surface of the legs. The curved shell material is 0.125" thick, with the density of carbon fiber/cyanate ester, but with 10 times the stiffness. The two flat discs are 0.2" thick, with densities set to represent the weight of carbon fiber/cyanate fiber that would occupy their volume plus another 35.3 pounds (16 kg) to represent the mirror/actuator assembly. The stiffness of the discs is also 10 times the stiffness of carbon fiber/cyanate ester.

Four basic variations on the leg design were analyzed: 2" x 6" hollow section; 2" x 10" hollow section; 1-3/8" x 10" hollow section; and 10" wide solid plate section. Note that the finite element models are comprised of plate elements, with material thickness assigned as a property. This means the position of the element in the model represents the midplane of the element. Therefore, for the hollow sections, the actual overall dimensions of the section are the lengths of the walls plus the wall thickness. For example, a 1-3/8" x 10" hollow section with 1/8" walls approximates a real section with outside dimensions of 1-1/2" x 10-1/8".

Within each variation of the leg design, weights and fundamental frequencies are determined for a variety of component thicknesses and material properties. In all the models, the materials comprising the legs and the buttresses are primarily carbon fiber/cyanate ester (K63712/CE-3 from COI Materials). Boron fibers are added to give the layup near zero coefficient of thermal expansion. Most of the analyses done are for a quasi-isotropic layup, which is modeled as an isotropic material. Some additional analyses are done using anisotropic layups to evaluate potential benefits from tailoring directional material properties.

Performance goals

For the purposes of this Engineering Study, the following performance goals are used:
-First mode natural frequency around 60-90 Hz or higher to minimize launch resonances.
-Adequate safety factors under quasi-static loading of 2.5 g's laterally, 5 g's axially or a combination of 2.5 g's laterally and 5 g's axially. Material failure is taken to mean Von Mises stresses exceeding yield stress for metals or maximum principal stresses exceeding tensile or the value of minimum principal stresses exceeding compressive strength for composite materials.
-Sufficiently high safety factors against buckling under quasi-static accelerations of 2.5 g's laterally, 5 g's axially or a combination of 2.5 g's laterally and 5 g's axially.

Analysis tools

Models were constructed and analyzed using ALGOR finite element analysis software. The analysis types used were: Linear Mode Shapes and Natural Frequencies; Linear Static Stress; Linear Critical Buckling Load; and Weight and Center of Gravity. All elements are either plate or thin composite type elements.

The Weight and Center of Gravity tool was used to find total weights for each model.

The Linear Mode Shapes and Natural Frequencies module was used to find fundamental natural frequencies.

The Linear Static Stress module was used to simulate quasi-static accelerations. Maximum and minimum principal stresses were found from individually applied accelerations of 5 g's in the x direction, 2.5 g's in the y direction, and 2.5 g's in the z direction, as well as from combined accelerations of 5 g's in the x direction and 2.5 g's in the y direction.

The Linear Critical Buckling Load module provided buckling factors of safety from the following loading conditions: 5 g's in the x direction; 2.5 g's in the y direction; 2.5 g's in the z direction; and 5 g's in the x direction combined with either 2.5 or –2.5 g's in the y direction, whichever produces lower a lower factor of safety. The results of these analyses are also expressed in terms of g loads producing critical buckling loads by multiplying the applied accelerations by the factor of safety result.

Limitations of the current model

The current models are for conceptual and comparative purposes only. The following assumptions, limitations and simplifications should be noted:

-No joining details are modeled—elements are assumed continuously joined along edges without overlapping tabs. No fasteners, inserts, or closeouts are modeled.
-No accounting is made for additional mass from MLI, coatings, or miscellaneous mechanical and electrical hardware, other than an estimate of 16kg for the mass of the mirror and its mounting and adjusting system.
-The legs and buttresses are modeled as mounting to a rigid surface—compliance of the Optical Bench is ignored.
-The mesh density of the models in this study is fairly coarse, and no effort has been made to increase the density and determine when the model converges. The current study is primarily for baselining and comparing construction configurations.

Analysis results

The focus of this study is on fundamental natural frequencies. Meeting the 60-90 hertz natural frequency goal and using near zero CTE materials practically ensures that other strength- and stiffness-related performance criteria are met.

The fundamental vibration mode of every design variation in this study is a twisting mode of the Secondary Mirror Assembly, where the Secondary Mirror Assembly rotates about the optical axis, and the legs are approximately in fixed/guided bending. Figure 2 illustrates this mode, viewed from the center of the field of view of the telescope. The frequency of this mode is sensitive to the mass moment of inertia of the Secondary Mirror Assembly about the optical axis. Minimizing this moment of inertia is desirable and should be considered when detailing the design of the assembly.

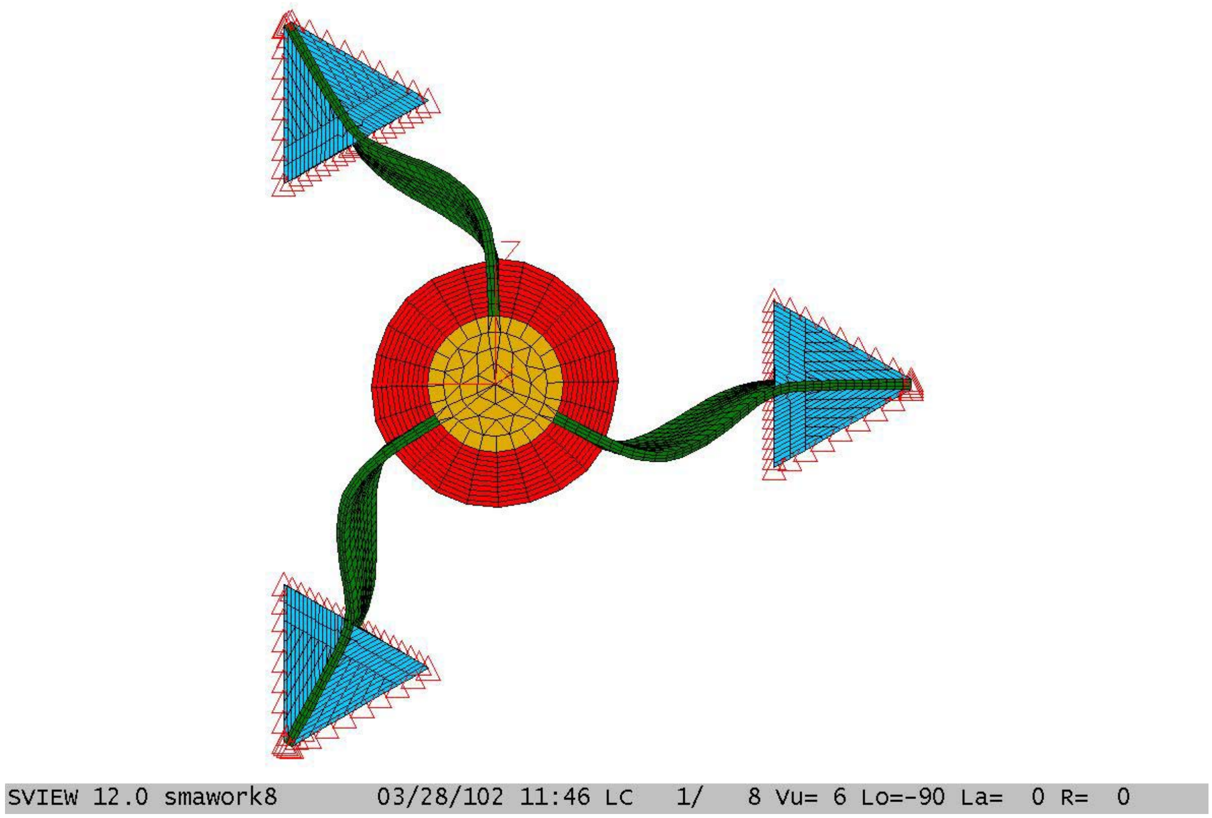

Figure 2-Typical first mode shape

A typical second vibration mode is illustrated along with the undisplaced shape in Figures 3 and 4. This is more of a translational mode, and may be excited to a greater extent than the rotational first mode. Note also that this mode shows significant deflection of the modeled Secondary Mirror Assembly, which is intended in this model to be very stiff. If the actual assembly can be made stiffer than this representation, the frequency of this mode will be higher than the results of this study, and conversely, if it is more compliant, this mode's frequency will be lower.

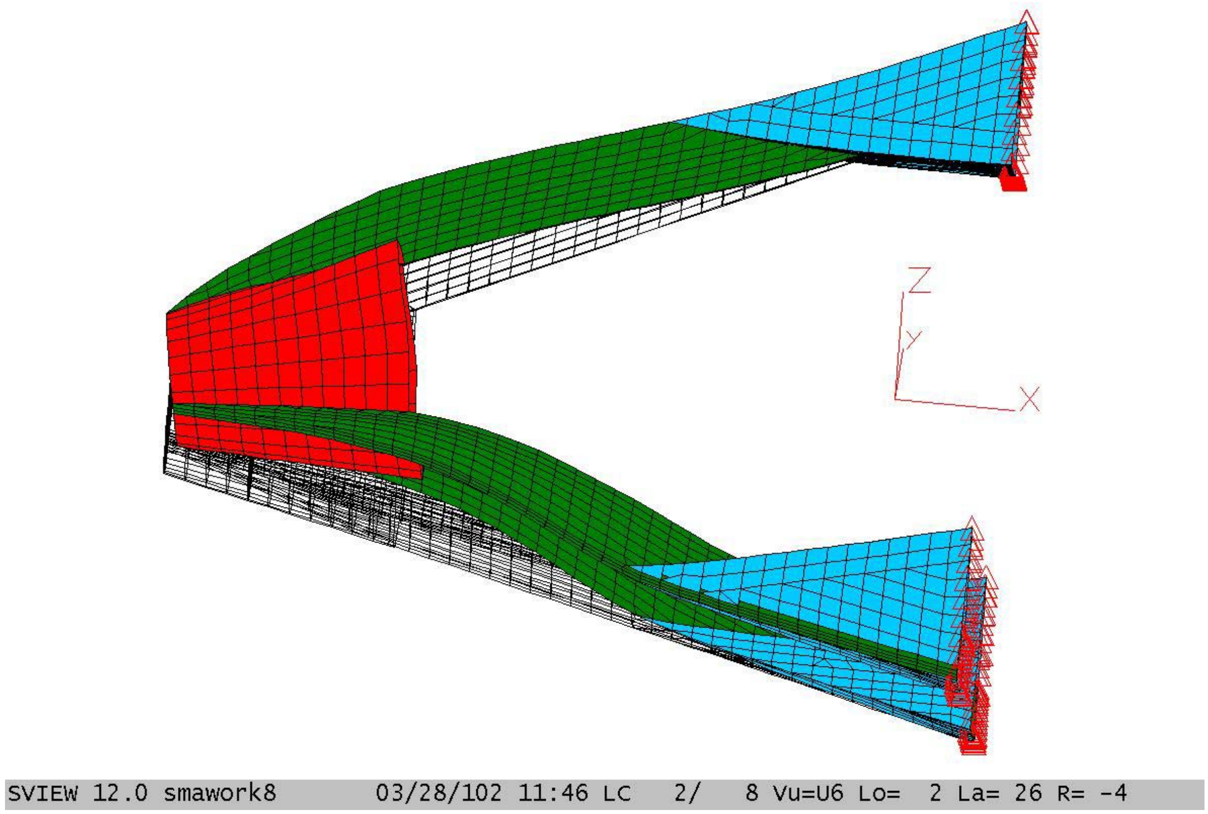

Figure 3-Typical second mode shape viewed approximately from one side

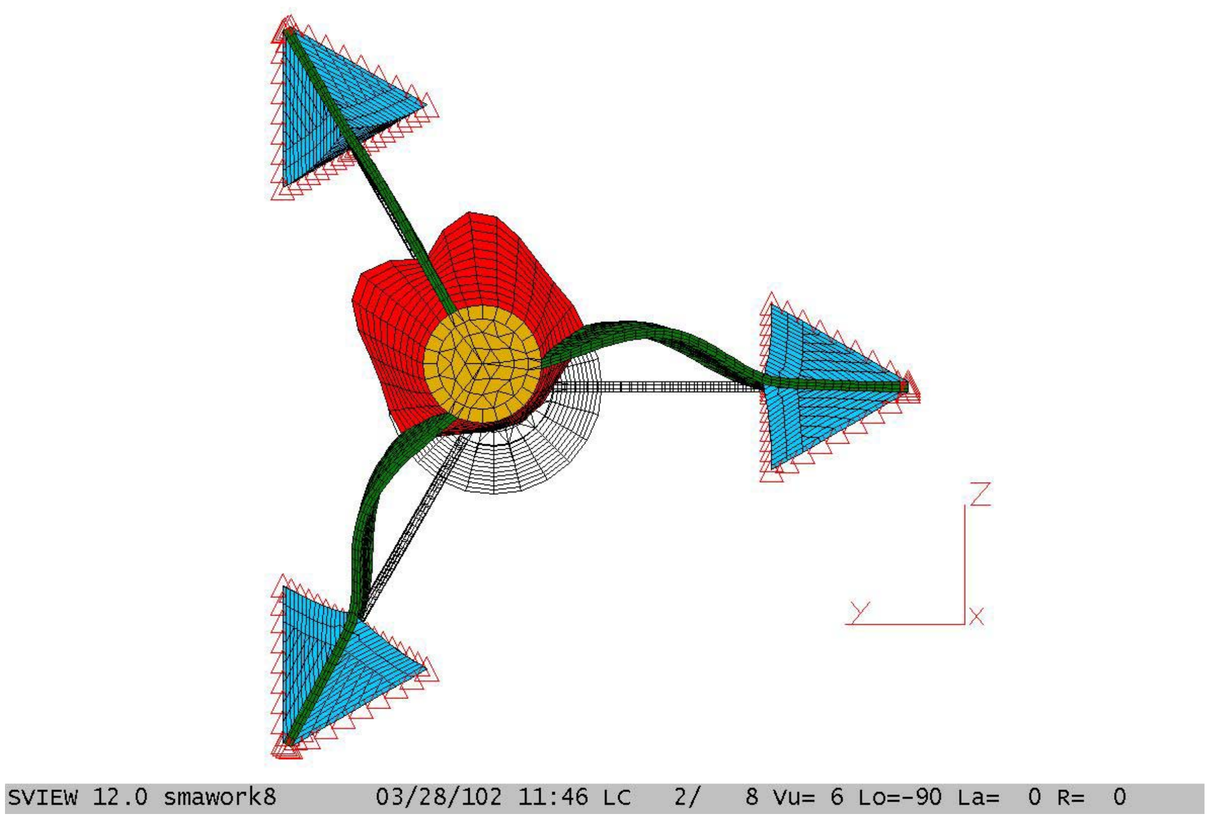

Figure 4-Typical second mode shape viewed from center of telescope field of view

Figure 5 shows how fundamental natural frequency varies with thickness of the walls of the legs for the three hollow leg configurations. These results assume the quasi-isotropic layup for both the buttresses and the legs, with 0.1" thick material for the buttresses. The total weight, including the Secondary Mirror Assembly, is indicated adjacent to each data point. Figure 6 illustrates the same relation for the flat-plate leg configuration, with quasi-isotropic layups and 0.1" thick buttresses.

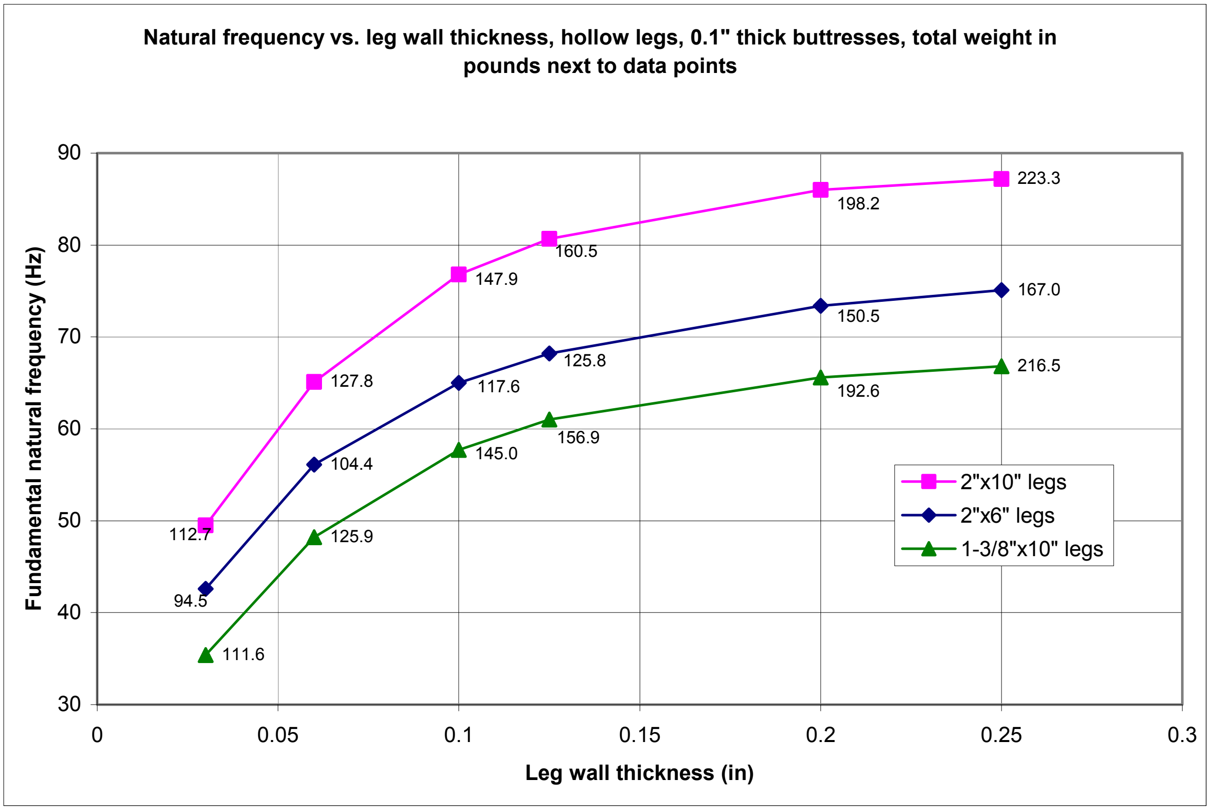

Figure 5

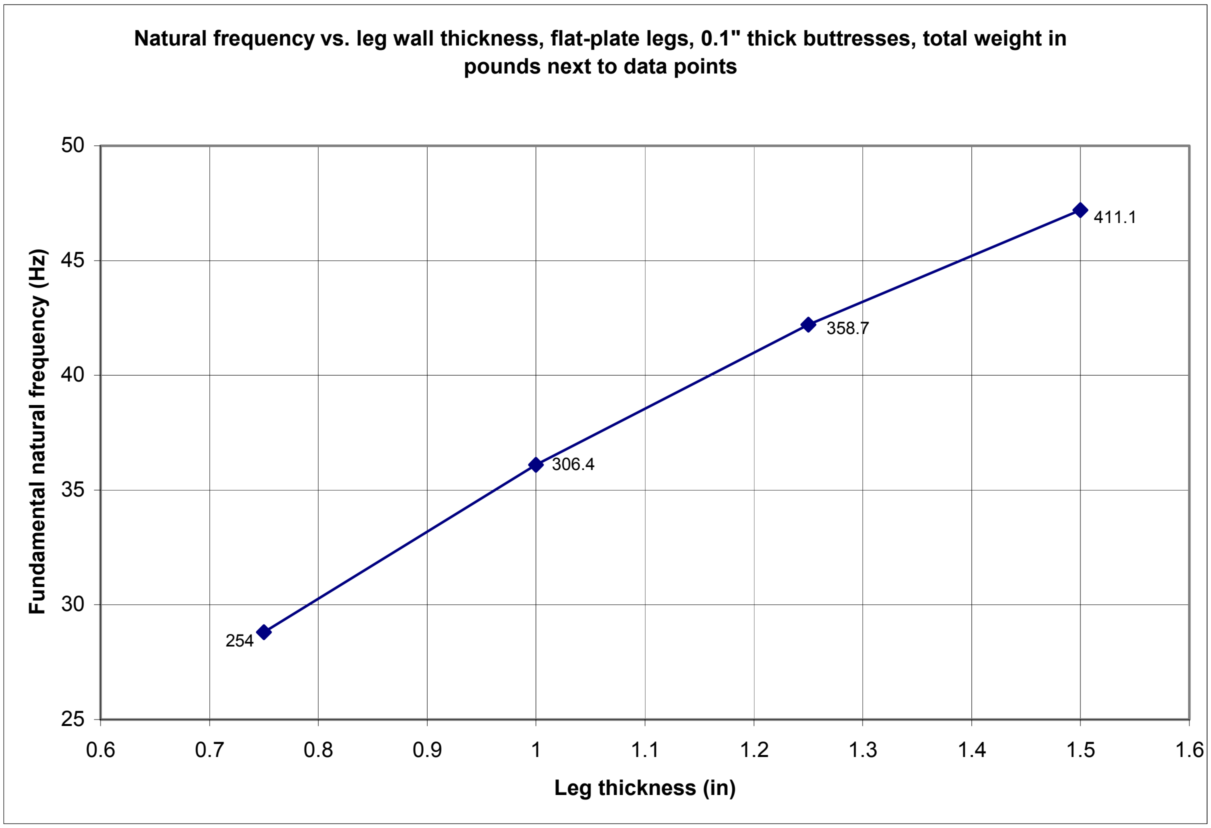

Figure 6

Figure 7 charts fundamental natural frequency against buttress material thickness for 1-3/8” x 10”, 0.125” thick legs with quasi-isotropic material. For buttress thicknesses less than approximately 0.05”, the lowest frequencies are oscillations of the buttress surfaces themselves. Again, the total weight is shown next to each data point.

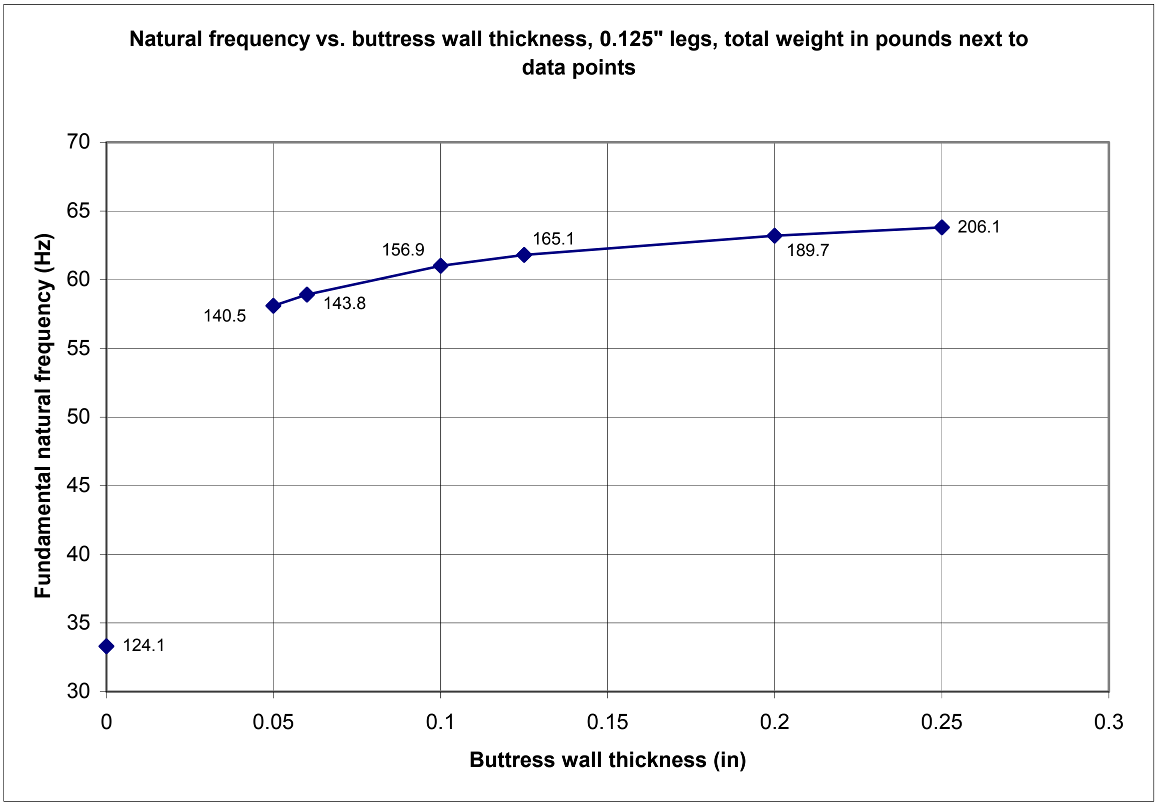

Figure 7

The bulk of the finite element work in this study has been on designs composed of the same, quasi-isotropic layup. This facilitates comparison of one design to another, independent of material properties. The use of fiber-reinforced materials, of course, allows for tailoring of material properties to suit the loading. To study the possible benefits of utilizing anisotropic layups, the design described in the Executive Summary section of this report was modeled using some different leg layups geared toward stiffening against first mode deflections. The greatest increase in first mode frequency seen in this study was using layups providing increased longitudinal stiffness for the 10" faces of the box section and increased shear stiffness for the 1-3/8" faces. The first mode for this configuration was 67 hertz, versus 61 hertz for the quasi-isotropic layup. Specifically, the 10" faces have moduli of 28.4 Mpsi longitudinal, 6.3 Mpsi transverse, and 7.2 shear, and the 1-3/8" faces have longitudinal and transverse moduli of 2.7 Mpsi and shear modulus of 15.1 Mpsi. The quasi-isotropic layup has longitudinal and transverse moduli of 20.0 Mpsi and shear modulus of 6.5 Mpsi. The described layup is not completely optimized, but studied layups with greater anisotropy yielded lower natural frequencies.

Another variation on the design in the Executive Summary was to increase the thickness of the 1-3/8" sides of the box section from 0.125" to 0.25". This change increased the total weight to 164.1 pounds from 156.9 pounds, but did not change the fundamental natural frequency.

The configurations already described in this report illustrate some of the trends and tradeoffs for the general tripod layout, employing a fairly simple leg design. Using straight legs as in these designs, however, pushes the bases of the legs outside the envisioned diameter of the baffle (~2.5m). This might require a larger base diameter or bulging features near the base of the baffle. Figure 8 shows a configuration with legs that kink at the front of the Primary Mirror to fit between the Primary Mirror and a 2.5-meter diameter baffle. The legs are buttressed using five flat surfaces. The illustrated design, with 1-1/2" x 10" x 0.125" thick legs and 0.2" thick buttresses weighs 170 pounds total.

This design's first two vibration frequencies are 62 hertz and 85 hertz, with mode shapes essentially the same as those previously illustrated. Under quasi-static loading of 5 g's axial and 2.5 g's transverse, the factor of safety against material failure is 14 and the factor of safety against buckling is over 100. Under one-g loading, of interest relative to optical performance during ground testing, the following deflections at the location of the mirror are predicted: under one g axially, axial deflection of 0.00006" (1.5 μm); under one g transverse (y or z), lateral deflection of 0.0014" (36 μm) and tilting relative to nominal of 0.0018° (6.5 arc-seconds).

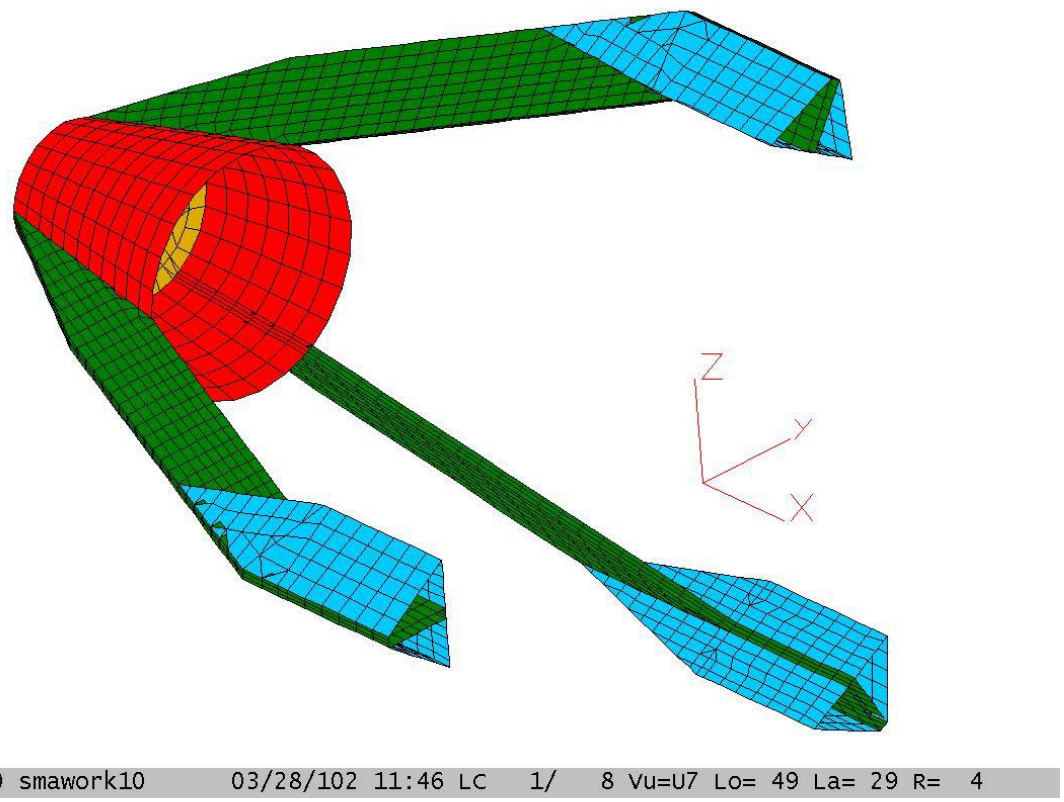

Figure 8-Configuration to fit within 2.5m diameter baffle